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Optimal design for a VLCC propulsion system based on torsional vibration analysis

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Abstract

For main engine changes, shafting torsional vibration of an exported VLCC tanker must be analyzed carefully. With the transition matrix method, the vibration models for the vessel with two different propulsion systems are made. The calculated results show there are forbidden rotated zones which are near to the vessel sailing speed, some necessary measures must be taken to make the forbidden rotated zones change in order that the vessel shafting system is to run safely in the long term instead of main engine changes.

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Keywords: VLCC; diesel; shafting; torsional vibration.

1. Introduction

In the designing process of a VLCC tanker which is exported to Ethiopia, the diesel type is changed from six cylinders to five cylinders. As the average pressures on crank shafts, where the activated forces are different, the torsional vibration of the propulsion system must be analyzed particularly.

Torsional vibration makes shafts broken where the maximum alternative stress takes place. If the vibration appears in shafting couplings, connected bolts will rupture sooner or later. When the alternative torque is larger than the gearboxes transferring permission, the following phenomenons appear such as corrosive teeth faces, teeth disjunction etc. In case there are high elastic couplings, the alternative torque makes their temperatures ascend and even burn. Simultaneously torque leads generators not to work stably, the machine body quivers strongly. The friction corrosions appear on the surface of propeller shaft cones. The generator power descends. If PTO equipments drive generators, the alternative torque moments make voltage fluctuate and incorporate in power network difficultly [1].

To avoid economic losses and security accidents, the forced vibration equations are written with the system matrix method according to different main engine types. The torsional vibration of the propulsion system is calculated and analyzed carefully. According to the calculated results, a reasonable propulsion shafting project is brought forward to reduce vibration and noises caused by the shafting torsional

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vibration. And the shafting running conditions and sailors' working environments are improved in the engine room.

2 The Shafting Torsional Vibration Model

The inertial mass system is adopted as the torsional vibration model. Thereinto concentrated masses are dealt with invariable inertias which are only with inertial moment and without elasticity. Two concentrated masses on the shaft are dealt with the connectors which are only with elasticity and without inertial moment [2-5]. The shafting inertial system is as following:

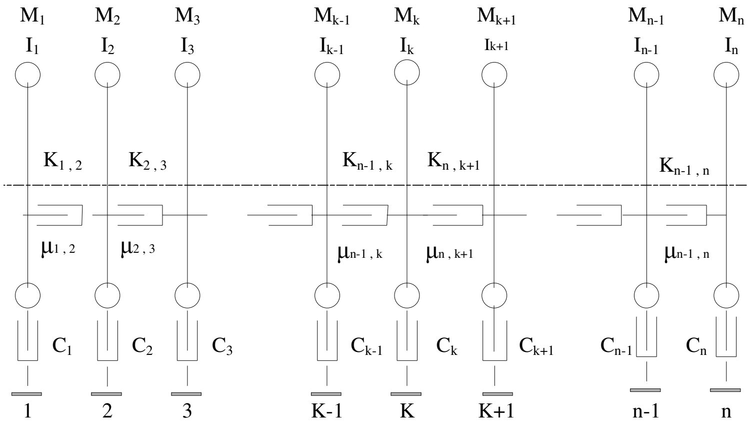

Fig.1 The shafting inertial system

According to a torsional system with n numbers of concentrated masses, the locomotion equations of the arbitrary mass K, which are effected by the inspirited torques M with the circularity frequency ω, are as following:

$$\begin{aligned}&I_K\ddot{\phi}_K + C_K\dot{\phi}_K + \mu_{K-1,K}\left(\dot{\phi}_K - \dot{\phi}_{K-1}\right) + \mu_{K,K+1}\left(\dot{\phi}_K - \dot{\phi}_{K+1}\right)\\ &+K_{K-1,K}\left(\phi_K - \phi_{K-1}\right) + K_{K,K+1}\left(\phi_K - \phi_{K+1}\right) = M_K e^{i(\omega t+\phi K)}\end{aligned} \tag{1}$$

Where:

φ_K, $\dot{\varphi}_K$, $\ddot{\varphi}_K$—the corner displacement, the corner velocity, the corner acceleration of mass K;

C_k —the damping factor of the linear mass K;

$\mu_{k-1,k}$, $\mu_{k,k+1}$ —the shafting damping factors at $K-1, K$ and K, $K+1$;

$K_{k-1,k}$, $K_{k,k+1}$ —the shafting torsional rigidity at $K-1, K$ and K, $K+1$;

I_k — the inertia of the mass K;

M_k — the breadth of the activated torque on the mass K;

φ_K — the original corner of activated torques;

ω — the round frequency of activated torques;

t — time.

The special results of locomotive equations are as following:

$$\varphi_K = A_K e^{i\varepsilon_K} \cdot e^{i\omega t} = \theta_K e^{iwt} \tag{2}$$

Where:

A_k — vibration breadth of mass K; ε_k — vibration corner of mass K; θ_k —plural breadth of mass K.

$$\theta_K = A_K \cos\varepsilon_K + iA_K \sin\varepsilon_K$$

Each mass expression is the same. According to the formulas (1) and (2), the numbers n of plural equations are got and solved with real number method. Then complex number is divided into real part and imaginary part, the numbers 2n of equations are got. The algebraic equation is calculated to get θ_k, A_k and adjunctive torques with the round frequency ω.

3 Torsional Vibration Analyses of Propulsion Systems with Different Engines

3.1 Torsional vibration analysis of the primary vessel

MAN B&W 6S50MC-C MkVII is selected as main engine for the primary vessel. The continual output power is 9480 kW, and the rotated speed is 127r/min. The vibration model of the propulsion system for the primary vessel is established by the transition matrix method, and the shafting inertial system is as following [6]:

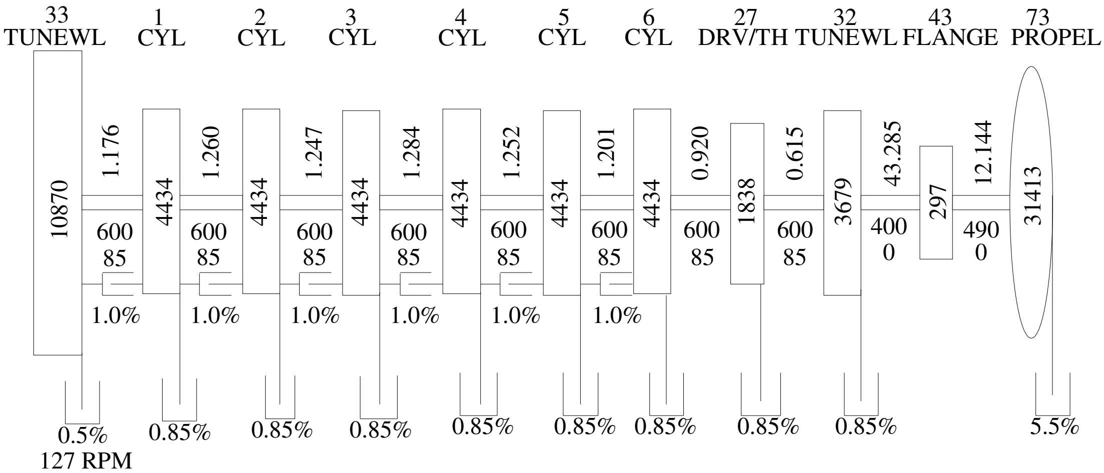

Fig.2 The shafting inertial system of the primary vessel

Diesel cylinders are dealt with concentrated masses, and they are rigid. The connection between each cylinder is flexible. And there is some frictional loss between each diesel cylinder. According to the shafting and propeller dimensions, the inertial number is to be gotten. The torsional torques and stress of the intermediate and propeller shafts are as following:

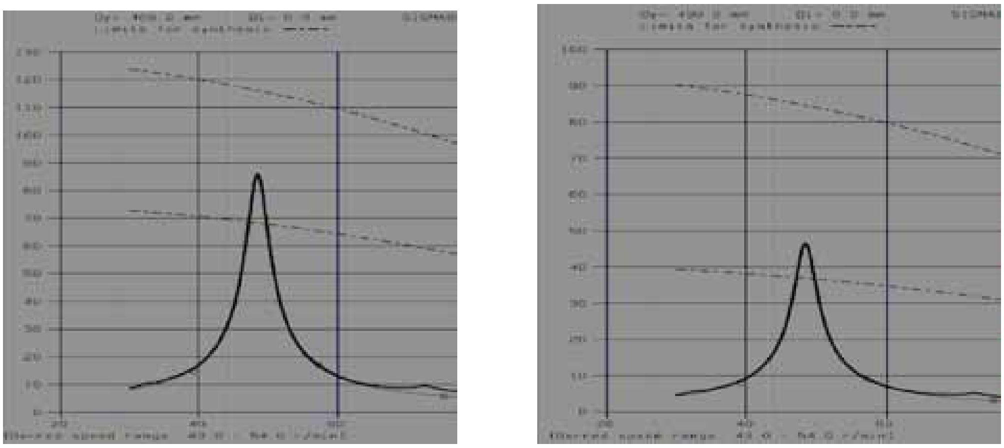

Fig.3 The torsional torques and stress chart of the intermediate and propeller shafts

On the basis of the above results, there is a forbidden rotated zone for the VLCC shafting. The shafting system must pass the forbidden rotated zone between 43 and 54 r/min quickly. Although the forbidden zone exists, it’s far away from the vessel sailing speed.

3.2 Torsional vibration analysis of the following vessel

MAN B&W 5S50MC-C8 is selected as main engine for the following vessel. The continual output power is 8300 kW, and the rotated speed is 127r/min. Because the propulsion power and two vessels scales are similar, the arrangement in the engine room is adjusted a little. Consequently the shafting system of the following vessel is almost the same with the original vessel. According to the mentioned method, the vibrational model of the propulsion system for the following vessel is made, and the shafting inertial system is as following:

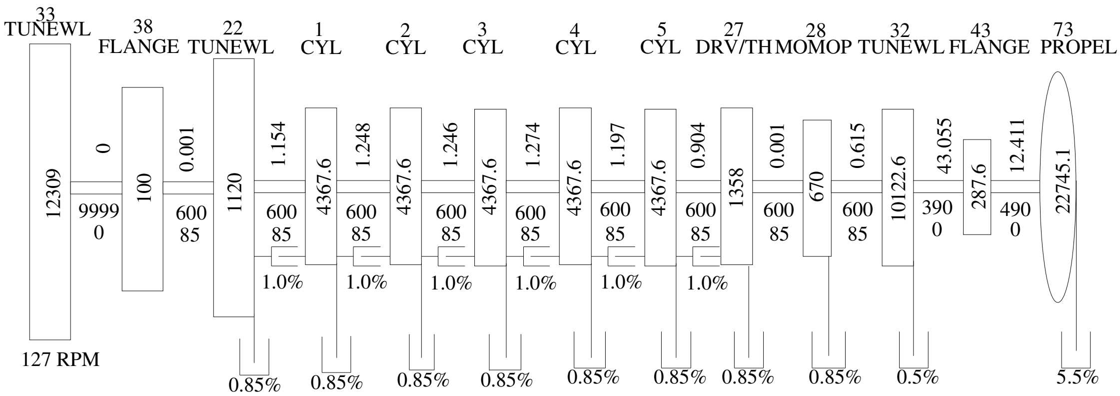

Fig.4 The shafting inertial system of the following vessel

According to the shafting and propeller dimensions, the inertial number is to be gotten in the same way. Because the arrangements in engine room are not to be changed greatly, the distance from the main engine wheel to the propeller central line is the same with the primary vessel. The propulsion shafting system of two vessels are almost the same. The torsional torques and stress of the intermediate and propeller shafts are as following:

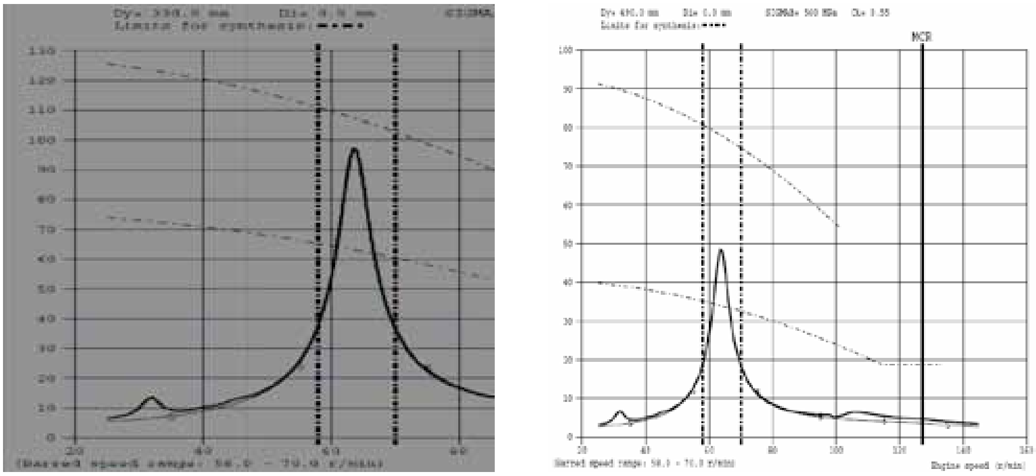

Fig.5 The torsional torques and stress chart of the intermediate shaft

According to the results, there is a forbidden rotated zone for the following vessel too. The shafting must pass the forbidden rotated zone between 59 and 70 r/min quickly. Because the forbidden rotated zone is near to the usual rotated speed of the main engine, the suitable flywheel should be installed to adjust the vibrational frequencies. The forbidden rotated zone moves to the left with the flywheel inertia changed.

4 Methods to Adjust Shafting Natural Frequency

For main engine changes, there are obvious differences in the torsional vibration results, although the shafting dimensions and arrangement are almost the same. For example, the torsional stress becomes larger and the forbidden rotated zone moves to the right. The torsional vibration characters of the following vessel shafting system can be improved with the following methods:

1 The shafting stiffness is adjusted the vibratory torques.

2 The high-carbon steel shafting with higher tensile strength is taken or the alloy steel shafting with high tensile strength is accepted. The shafting diameters and stiffness are reduced, and the shafting allowable stress is improved.

3 According to different inertial flywheels, heavy flywheels are chosen to reduce the shafting torsional stress.

4 The flywheels are installed in the diesel free end to reduce the shafting torsional stress.

5 The suitable dampers are selected to avoid the forbidden rotated zone and improve the vessel manoeuvrability.

6 The propeller should be installed in the optimal corner and adjust the intersection angle between the central line of propeller blades and the diesel crankshaft. The phasic difference, which is caused by the propeller activated torques, counteracts the activated torque which is caused by the diesel to reduce the shafting torsional stress.

7) If the diesel firing order is permitted to be adjusted, the forbidden rotated zone can be avoided.

5 Conclusions

With the transition matrix method, the shafting system model of a VLCC oil tanker, which will be exported to Ethiopia, is analyzed. According to the calculated results, the propulsion system must be adjusted with some methods to change the forbidden rotated zones so that the zones are far away from the vessel's sailing speed. When diesels are selected as main engines for vessels, the shafting torsional vibration must be analyzed particularly in order that the shafting system works onboard safely for a long time.

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